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(71) Applicants (for all designated States except US): **MÆRSK CONTAINER INDUSTRI A/S** [DK/DK]; Bjerndrupvej 47, DK-6360 Tinglev (DK). **LODAM ELEKTRONIK A/S** [DK/DK]; Grundvigs Allé 163, DK-6400 Sønderborg (DK).

(72) Inventors; and
(75) Inventors/Applicants (for US only): **RAUN, Niels, Peter** [DK/DK]; Gamle Kongevej 19, DK-6200 Aabenraa (DK). **ANDERSEN, Jens** [DK/DK]; Rugtoften 7, Guderup, DK-6430 Nordborg (DK). **MINDS, Gunnar** [DK/DK]; Lykkenshøj 64, DK-8220 Brabrand (DK).

(74) Agent: HOFMAN-BANG ZACCO A/S; Hans Bekkevolds Allé 7, DK-2900 Hellerup (DK).

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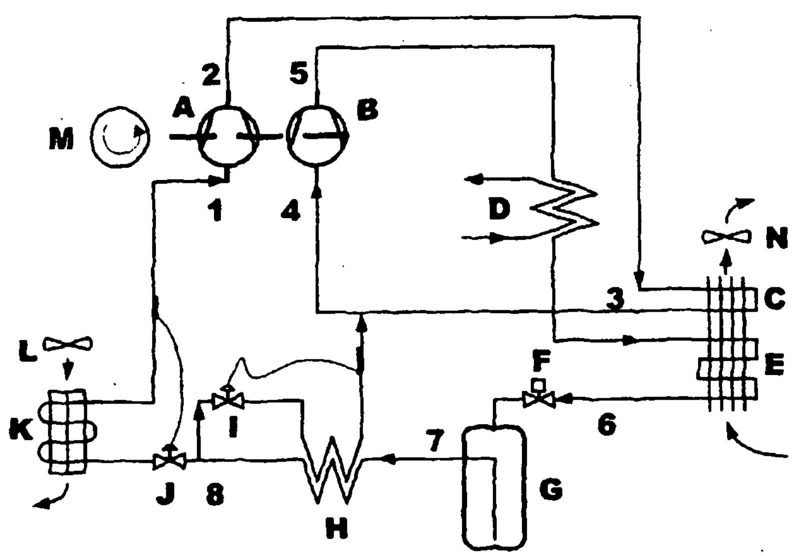
(54) Title: A VAPOUR-COMPRESSION-CYCLE DEVICE



(57) Abstract: The present invention involves a method and a device for regulating the capacity of a vapour-compression-cycle that comprises a compressing device, a condenser, at least one cooler, a variable-flow control means, and an evaporator, connected in series and forming an integral closed circuit with a refrigerant that operates under trans-critical conditions, where in the capacity is regulated by regulating the number of revolutions of the drive shaft of the compressing device as a function of the required capacity; and that the variable flow control means is regulated as a function of the temperature and/or pressure of the refrigerant measured at the high pressure side of the system with respect to optimizing the COP. Hereby it is optained to provide a novel, improved, efficient and simple means for regulating the capacity of a trans-critical vapour-compressing-cycle device avoiding the above shortcomings; and to improve the efficiency of the vapour-compression-cycle process.

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A vapour-compression-cycle device

The present invention relates to vapour-compression-cycle devices, such as refrigerators, air-conditioning units, heat pumps, etc, using a refrigerant operating in a closed circuit under trans-critical conditions, and specifically to the methods of regulating the capacity of such devices.

Conventional vapour-compression-cycle devices use refrigerants (as for instance R22, R134a, R404a, CF_2Cl_2) operating entirely at sub-critical pressures. A number of different substances or mixtures of substances may be used as refrigerant. The choice of refrigerant is among others influenced by condensation temperature, as the critical temperature of the fluid sets the upper limit for the condensation to occur. This technology is treated in full details in the literature, e.g. in the Handbooks of American Society of Heating, Refrigerating and Air Conditioning Engineers Inc., Fundamentals 1989 and Refrigeration 1986.

The operation of a vapour-compression-cycle device under trans-critical conditions have been practiced to some extent. Until the use of halocarbons took over in the fifties, CO_2 (Carbon dioxide) was a generally known refrigerant. CO_2 refrigeration technology is described in earlier literature, e.g. P. Ostertag "Kälteprozesse", Springer Verlag 1933 or H.J MacIntire "Refrigeration Engineering", Wiley 1937.

Vapour-compression-cycle devices operating under trans-critical conditions are known from German patent 278095 from 1912. This patent describes a method involving two-stage compression of a refrigerant, with intercooling of said refrigerant in the supercritical region. Two-stage compression involving intercooling is ideal when the process employs a cooling agent that operates under trans-critical conditions as the temperature generated under the compressing process would reach high levels and influence the performance of the cycle. However, the method described suffers from the drawback that the

components are placed in a manner that does not comply with today's demands for high efficiency products/processes.

A further vapour-compression-cycle device operating under trans-critical conditions is known from patent application No. PCT/NO89/00089 that describes a vapour-compression-cycle device and wherein the capacity regulation of said device employs a single-stage compression cycle process comprising a single-speed compressing device, where the capacity is regulated by varying the refrigerant-enthalpy difference in the evaporator by changing the specific enthalpy of the refrigerant before throttling. This modulation of specific enthalpy is achieved by varying the pressure in the high-pressure side of the compression cycle device that adjusts the throttling means opening.

It is generally known that the commonly used refrigerants (halocarbons), when employed in a vapour-compression-cycle device, usually ensures a high COP (coefficient of performance) for the device in question. The COP (coefficient of performance) is defined as the ratio between the power (kW) consumed for driving the compressing devices and the cooling capacity (kW) produced by the vapour-compression-cycle device. For vapour-compression-cycle devices employing for instance R134a, R404a, a ratio of 1:3 - 4 is not uncommon. For vapour-compression-cycle devices employing CO_2 the COP is approximately 1:2-3.

Due to the ozone-depleting effects of the commonly used refrigerants (halocarbons) international actions have been taken to reduce or prohibit the use of these gasses. As a consequence, there is a need to find alternatives to the known technology that are environmentally friendly and still economical in all aspects.

It is therefore an object of the present invention to provide a novel, improved, efficient and simple means for regulating the capacity of a trans-critical va-

pour-compression-cycle device avoiding the above shortcomings; and to improve the efficiency of the vapour-compression-cycle process.

Another object of the present invention is to provide a vapour-compression-cycle device employing a refrigerant that operates under trans-critical conditions and with the smallest possible power consumption; and to employ attractive refrigerants with respect to environmental hazards.

The above and other objects of the present invention are achieved by providing a method operating at usually or usual trans-critical conditions, where the capacity is regulated by regulating the number of revolutions of the drive shaft of the compressing device as a function of the required capacity; and wherein the variable flow control means is regulated as a function of the temperature and/or pressure of the refrigerant measured at the high pressure side of the system with respect to optimizing the COP (coefficient of performance) of the system.

Another object of the present invention is achieved by providing regulation means for the vapour-compression-cycle device that comprises means for regulating the revolutions of the drive shaft of said compressing device for regulating the capacity; and means for controlling said variable control means as a function of the temperature of the refrigerant on the high pressure side with respect to optimizing the COP of the system.

Still another object of the present invention is to provide a control means, said control means for the conditions given regulating the throttling means on basis of temperature/pressure of the refrigerant on the high-side for optimizing the COP (coefficient of performance)

Yet a further object of the present invention is to provide a vapour-compression-cycle device comprising two or more compression stages. By introducing two or more compression stages, it has been proven that an accurate regulation is possible by utilizing more simple specifications of the algorithm for the optimal pressure after the variable flow control means.

A further object of the present invention is to provide a method for regulating the regulation means for the variable flow control means at decreasing temperature of the refrigerant arranged to regulating said variable flow control means until a temperature/pressure inferior to the critical temperature, whereupon the opening for refrigerant of the variable flow control means is fully opened, and at increasing temperatures the regulation means starts to regulate said variable flow control means at a temperature higher than the critical temperature.

By this method it is ensured that it will be possible to control and regulate the variable flow control means at temperatures of the refrigerant/gas about the critical value of ca. 32°C in order to prevent to many switching in/off's of the compressing devices.

A further object of the present invention is to provide a method, wherein the refrigerant, after the first compression stage and before the second compression stage, is cooled in a first cooler that employs air as coolant, said first cooler being with respect to the airflow placed in series, and after a second cooler that employs air as coolant for cooling/condensing the refrigerant after the second stage compression.

Yet a further object of the present invention is to provide a method for regulating a transcritical vapour-compression-cycle device employing CO_2 (carbon dioxide) as refrigerant.

Yet another object of the present invention is to provide a device comprising means for measuring the temperature of the refrigerant on the high-pressure side with respect to optimizing the COP of the system as a function of the temperature or pressure.

Yet another object of the present invention is to provide a vapour-compression-cycle device comprising a cooler employing water as coolant arranged in the integral closed circuit after a compressing device and before the air-cooled cooler.

By an arrangement with the suggested configuration, it is achieved that the second cooler (E) that employs air as coolant does not transfer heat from the circulation air and on to the first cooler (C) that employs air as coolant. Due to a high temperature of the circulation air, the heat transmission and by this the capacity of the first cooler would then approach a zero effect.

Still another object of the present invention is to provide a vapour-compression-cycle device comprising a receiver arranged in the integral closed circuit after the variable flow control means but before the economizer. With the suggested configuration, it will be achieved that the receiver accumulates the refrigerant in liquid state. The pressure elevated in the receiver will at any time be the pressure generated when the refrigerant expands adiabatically from the gas cooler to the threshold curve.

Another object of the present invention is to provide a vapour-compression-cycle device employing CO_2 (carbon dioxide) as refrigerant by which a refrigerant operating under trans-critical conditions is environmentally friendly.

Yet another object of the present invention is to provide a vapour-compression-cycle device comprising coupling means for establishing water cooling, by which it will be made possible to establish water-cooling when the container is loaded onboard a ship, and air cooling of the heat exchangers/condensers is not sufficient for cooling the refrigerant/gas.

A further object of the present invention is to provide a vapour-cycle-compression device comprising means for coupling to a ship's electrical power system, and wherein the compression device is driven by an electrical motor means coupled to said electrical power system, by which it is ensured that the vapour cycle compression device is operational when loaded on a ship for transportation.

Yet another object of the present invention is to provide a method employed for refrigerating the interior of a refrigerated container. By this it has been achieved that goods requiring a refrigerated atmosphere can be transported/stored in a container that provides such refrigerated atmosphere.

Brief description of the drawings

The invention will now be described in brief terms in the following with reference to the drawings, Figure where:

Figure 1 is a graph illustrating the vapour compression cycles for the refrigerants R134a and CO_2 (carbon dioxide); and

Figure 2 is a graph illustrating the capacities (Q_0) dependency of the temperature of the refrigerant gas/vapour at output of the cooler at maintained pressure;

Figure 3 is a graph illustrating the capacity (Q_0) dependency of the pressure in the cooler for maintained gas temperature;

Figure 4 illustrates calculated pressure in the cooler for optimal COP depending off the evaporation temperature and gas temperature at output of the cooler;

Figure 5 illustrates calculated pressure in the cooler for optimal COP depending off on evaporation temperature and the gas temperature at the output from the cooler;

Figure 6 illustrates calculated sensitivity for correctly selected pressure in the cooler for optimal COP (T0 = -25 °C);

Figure 7 is a graph illustrating an H-log P diagram for at two-stage system operating at trans-critical conditions;

Figure 8 is a graph illustrating a H-log P diagram for at two-stage system operating at sub critical conditions;

Figure 9 is a calculated comparison of gas temperature; and

Figure 10 is a diagram for determining the adjustment pressure; and

Figure 11 is a schematic representation of a trans-critical vapour-compression-cycle device according to a preferred embodiment of the invention.

In the figures letters refer to items described and numbers refer to the various stages of the refrigerant in an enthalpy/temperature diagram, in the following figures.

When CO_2 is used as refrigerant in vapour-compression-cycle device or heat pumps, the process will depend on the temperature at the heat exchanger/condensing unit being above or below the critical temperature. At temperatures of the coolant in the condenser/heat exchanger below the critical value, the integral circuit process of the refrigerant (CO_2) will not be different from the integral circuit process of other refrigerants complying to the Carnot cycle/process. However, at temperatures higher than the critical value, the integral circuit process will be different as CO_2 cannot condense at temperatures above 31 °C. This does not mean the process cannot supply cooling or heating, only that the process is to be configured to comply with another integral circuit process, namely the Lorentz proces. The condenser will no longer be used for condensing the refrigerant, but solely for cooling of the trans critical coolant/fluid, and is hence described as the "gas-cooler"

In figure 1 the trans critical integral circuit process is illustrated in comparison to the conventional integral circuit process employing R134a as refrigerant. Listed in a H,log(P)-diagram (H for enthalpy, P for pressure). In the figure both isotherms (40 °C) for both refrigerants – CO_2 (carbon dioxide) and R134a) her mangler noget. Both processes operate at an evaporating temperature at minus 10 °C and up against an outdoor temperature or a water temperature of the coolant of approximately 40 °C. As it will appear from Figure 1, the integral circuit process employing CO_2 operates at a pressure lever higher than the integral circuit process employing R134a. This means that pipe systems, receivers, and components in general are to be configured for this much higher operating pressure. It will further appear from Figure 1 that the expansion vessel/throttling means have supercritical fluid at the inlet where it normally for R134a will be a liquid. This means that the liq-

uid formation for the trans critical process is taking place in the expansion vessel/throttling means during the expansion though the nozzle, which normally do not give reason for problems

At operation at trans-critical operations, there is no connection (dependency) between pressure and temperature. This means that the system has been awarded yet another degree of freedom, in the sense that it will be possible to control/regulate the pressure in the gas-cooler and the temperature independently.

Figure 2 illustrates how the cooling capacity is increased dramatically by changing the temperature of the refrigerant (CO_2) by discharge from the gas cooler. The work of the compressing device (W_k) remains the same. Under normal conditions the temperature of the CO_2 at output of the gas-cooler cannot be chosen arbitrarily, but will be dependent of the conditions of the integral circuit process device given. This may be the temperature of the air, when air is employed as cooling agent, or the temperature of the cooling water, when water is employed as cooling agent. Since it is not possible to control the temperature of the refrigerant at output of the gas cooler, this means that instead a pressure optimal for a requested run (low/high capacity) or just for the conditions given, to maximize COP (coefficient of performance) for the integral circuit process device. In some cases it may be relevant to reduce the flow of coolant (air/water) over the gas cooler/condenser in order to reduce the capacity. The result may then be that the COP (coefficient of performance) of the integral circuit process device is not optimal; however, these steps will prevent many start/stops of the integral circuit process device.

Capacity regulating of a trans-critical integral circuit process device employing CO_2 as refrigerant can be achieved –apart from the normal methods – by controlling the pressure in the gas-cooler. For a temperature of the CO_2- refrigerant given at output of the gas cooler, (determent by exterior conditions)

it is thus possible to regulate the capacity of the integral circuit process device within a wide range. The principle is illustrated by Figure 4.

Assuming the temperature of the refrigerant (CO_2) at discharge of the gas cooler/heat exchanger is determent by the exterior conditions and has an value of 35 °C as will appear from Figure 3, it will then be possible to increase the capacity of the integral circuit process device considerably by raising the pressure in the gas cooler/heat exchanger slightly (en smule). The other way around, it will be possible to reduce the capacity of the integral circuit process device should this be required, where at the same time the energy consumption of the compressing device is reduced and a energy retrenchment is obtained.

There is often no need for this type of capacity regulation or it may be achieved by other means, for instance by capacity regulations of the compressing device (cylinder switch-off or regulation of number of revolutions) In this situation it will be possible to regulate in respect to the maximal COP of the process.

Figure 4 illustrates the calculated optimal choice - in respect to maximal COP – of the gas pressure in the gas cooler/heat exchanger dependent of the temperature of the refrigerant gas at output from the gas cooler. As it will appear from Figure 4 the optimal high pressure is dependent of both the gas temperature at output of the gas cooler, but even of the evaporation temperature. Further the optimal pressure will depend on the mechanical efficiencies of the compressing devices and pressure losses in exchangers and pipes. On the background of an analysis, the optimal high-pressure can be found on the basis of varying evaporation temperatures (T_e) and temperatures at output of the gas cooler ($T_{g,2}$). The isentropic efficiency compressing device is kept at 0,6.

Figure 5 illustrates calculated pressure in the cooler for optimal COP dependent of evaporation temperature and the gas temperature at the output from the cooler. On this basis a mathematical formula has been expressed, stating the optimal pressure of the gas/refrigerant at output of the cooler, for optimal COP on the basis of T_e and $T_{g,2}$.

$$P_{g,2}\,(\text{optimal}) = 0{,}7244 - 0{,}275 * T_e + 2{,}275 * T_{g,2}$$

In figure 6 the sensitivity for correct chosen high-pressure is illustrated. As it will appear from the curves, choosing a operating pressure below the optimal pressure as this result in a low COP. The slope is relatively high at these pressures. It will however appear that for pressures higher than the optimal operating pressure, the slope is low. The optimum of each curve represented by a condensing temperature is connected by a broken line in their vertex.

Figure 7 is a graph illustrating a H-log P diagram for a two-stage system operating at trans-critical conditions. The numbers are referring to the stages of the circle process, where: (1) is the condition of the refrigerant at the inlet of the first compressing device (A). (2) the condition after the compression by the low-pressure compressing device (A). the condition of the refrigerant at (3) is after cooling of the gas in the first cooling device (C) and 4 is the condition at the inlet of the high-pressure compressing device (B). The pressure is then elevated to the pressure at (5) after compression in the high-pressure compressing device (B). Hereafter the refrigerant is cooled reaching the stage at (6) after cooling/condensation in the gas cooler/condensing unit (E). In the receiver the pressure of the refrigerant is reduced at the stage with number (7). Hereafter the refrigerant is cooled further by the economizer, by which some sub-cooling is ensured reaching the stage (8), which indicates the condition of the refrigerant before the evaporating in the evaporator.

However, for two-stage vapour compression cycle devices plants comprehensive calculations have shown that it is possible to obtain sufficient accu-

racy by use of a more simple description of the algorithm for the optimal gas pressure after the variable control means. See Figure 10.

Calculations have been made for the suction pressure on 10, 20 and 30 bar, respectively, which is the maximal variation for the suction pressure in a vapour compression cycle device for transportation purposes. As is the case when one-stage vapour-compression-cycle devices are concerned, calculations show that the optimal depends on the suction pressure, but the dependency is less severe than that of one-stage devices. This is due to the fact that the suction pressure for the second compressing stage varies only half as much as for the first stage. Therefore, the deliberations have been whether it is possible to replace the fairly complicated expression with a simpler one and use a curve that is independent of suction pressure. When the curve is used for a 20-bar suction pressure, it has been found that the theoretical mistake made that the curve is used for a 20-bar suction curve only is less than 1% at a gas temperature of 50°C, decreasing to zero at the critical pressure of 32°C where all the curves collapse. In practice the gas temperature will never exceed 45°C.

It is worth noticing that the variable flow control means continues to be a controlling factor for the gas temperature, decreasing until 27.5°C, following which it must be completely open. Conversely, at increasing gas temperature a regulation is not initiated until the temperature exceeds 32.5°C. This function is necessary to avoid too excessive switch in and switch outs at gas temperatures around the critical temperature of about 32°C.

Calculation of the optimal gas pressure as disclosed in Figure 11 have included the values for isentropic efficiency found by tests on current vapour-compression-cycle devices engaging CO2 as refrigerant.

It is a further characteristic that gas pressure and temperature within the receiver will always settle at a level that corresponds to the saturation pres-

sure that prevails where the gas expands, at constant enthalpy, above the throttle valve to upper or lower threshold curve.

Figure 11 shows a schematic representation of a trans-critical vapour-compression-cycle device according to a preferred embodiment of the invention, comprising two-stage compression of a refrigerant. In the integral closed circuit the vapour-compression-cycle device comprises a first compressing device (A) with an inlet and an outlet side, connected in series to a first heat exchanger (C) and employing air as coolant; and a subsequent second compressing device (B) with an inlet and outlet side, connected to a counter flow heat exchanger (D) and employing water as coolant. In the closed circuit after said counter flow heat exchanger (D), a second heat exchanger (E) employing air as coolant is arranged. Said first and second heat exchangers (C, E) employing air as coolant are, with respect to the airflow, arranged in series with the second heat exchanger (E) first and the first heat exchanger (C) second, arranged in a trunk-like or similar space providing a air path for the cooling air. Hereafter a ventilator fan is arranged to ensure a sufficient airflow for the heat exchanging process. In the integral closed circuit after the heat exchanger (E), a sensor device followed by a variable flow control means (F) comprising regulation means (not shown) and a receiver (G) are arranged, followed by an economizer (H) with an inlet and outlet, and a throttling means (expansion vessel) (J) comprising a sensor attached to the suction line, described in the latter. In the integral closed circuit, an economizer by-pas line comprising an inlet and outlet is connected with entrance after the economizer (H) and before the throttling means (expansion vessel) (J) and the outlet connected to the integral closed circuit after the first heat exchanger (C) and before the second compression device (B). In the economizer by-pass line, a throttling means (expansion vessel) (I) is arranged before the economizer, said throttling means (expansion vessel) (I) with the sensor being attached to the in the economizer by-pass line after the economizer (H). The function of throttling means (expansion vessel) (I) will be known to the skilled person, hence it will not be described in further

detail in the following. it is the function of the economizer (H) to ensure some degree of sub-cooling of the refrigerant. In the closed circuit after the throttling means (expansion vessel) (J) an evaporator (K) with an inlet and an outlet is connected to the inlet connected with the throttling means (expansion vessel) (J) and the outlet to the side connected to the inlet of the first compression device (A), by a line/pipe hereafter described as suction line. The function and arrangement of throttling means (expansion vessel) (J) and sensor will be known to the skilled person, hence it will not be described in further detail in the following. It will be understood that the evaporator (K) is installed inside a container in which a refrigerated atmosphere is required as it will be likewise understood that a ventilation fan (L) is advantageously arranged to circulate air around the evaporator (K).

The first compression device (A) and the following second compression device (B) both comprising a drive shaft are, in the representation of Figure 1, shown as a two-stage compressing device unit, sharing the same drive shaft, said drive shaft being coupled to a motor means. It will, however, be understood that the above described embodiment of the invention is intended to be exemplary only and not in any way limiting the scope of the invention. It will be appreciated that the two-stage compressing device driven by one motor means can be replaced by two single stage compressing devices each comprising a motor means.

The vapour-compression-cycle device comprises a first sensor means arranged on the high-pressure side in the closed circuit before the variable flow control means (F) and after the second cooler (E). Another sensor means is arranged in the closed circuit after the evaporator and before the first inlet of the compression device (A). The sensor means is, in a preferred embodiment, temperature sensors, monitoring the temperature condition of the refrigerant/gas and giving a input signal to the vapour-compression-cycle devices regulation means for regulating the capacity of the vapour-compression-cycle device.

The vapour-compression-cycle device according to a preferred embodiment of the invention is suitable for installation in a container of the type used on in particular ships, for the transportation of goods that require refrigerated surroundings. When the device is loaded onboard, the ship's power system is connected to provide the drive motor for the compressing devices (A, B) and all other electrical components that are powered electrically, and the counter flow heat exchanger (D) is connected to a cooling water system, said cooling water system providing cold water for containers equipped with a vapour-compression-cycle device comprising a heat exchanger employing water as cooling agent/coolant. After connection to said systems, the ventilation fan (N) is turned off and the cooling of the refrigerant after the second compression device (B) is taken over from the second heat exchanger (E) by the counter flow heat exchanger (D). An airflow as the result of convexion in the airpath ensures some amount of cooling air or the first heat exchanger (C)

The practical problem occurs when the same vapour-cycle-compressing-device is set to handle both super – and sub-critical pressure. The regulation function must be capable of handling this conversion. At sub critical operation, the variable control means, in principle must be fully open and in super critical operation adjustable. It will however be practical possible to make this conversion pass without dramatic variations in the temperature of the refrigerant.

The regulation of the variable flow control means is illustrated in figure 9, listing the comparison between the gas temperature at output gas cooler and optimal gas pressure, and with the regulating curve sketched, said regulation curve determent by the following equation.

a = 282.05394

b = -1.5741522

c = -167064.7

d = 425285.17

e = -3.68426 x 109

$$P_{reg} = a + b \cdot T_{g,2} + c \cdot \frac{\ln(T_{g,2})}{T_{g,2}^{2}} + \frac{d}{T_{g,2}^{2}} + e^{-T_{g,2}}$$

For T_{g2} smaller than 20 °C the variable flow control means is kept open. For $T_{g2} < 20$ °C, then $T_{g2} = 20$ °C.

Claims

1. Method for regulating the capacity of a vapour-compression-cycle device that comprises a compressing device, a condenser, at least one cooler, a variable-flow control means, and an evaporator, connected in series and forming an integral closed circuit with a refrigerant that operates under trans-critical conditions, **characterized in** that the capacity is regulated by regulating the number of revolutions of the drive shaft of the compressing device as a function of the required capacity; and that the variable flow control means is regulated as a function of the temperature and/or pressure of the refrigerant measured at the high pressure side of the system with respect to optimizing the COP.

2. Method according to claim 1 **characterized in that** a control means for the conditions given regulates the variable flow control means on the basis of temperature/pressure of the refrigerant on the high-side for optimizing the COP.

3. Method according to claim 1 or 2, **characterized in that** that the compression comprises two or more compression stages

4. Method according to claim 3, **characterized in that** the regulation means for the variable flow control means at decreasing temperature of the refrigerant is arranged to regulate said variable flow control means until a temperature/pressure inferior to the critical temperature is reached, whereupon the opening for refrigerant of the variable flow control means is fully opened; and the regulation means at increasing temperatures starts regulating said variable flow control means at a temperature higher than the critical temperature.

5. Method according to claim 3 or 4, **characterized in that** the refrigerant, after the first compression stage and before the second compression stage, is cooled in a first cooler employing air as coolant, said first cooler with respect to the airflow being placed in series, and after a second cooler employing air as coolant for cooling/condensing the refrigerant after the second stage compression.

6. The method according to one of the preceding claims, **characterized in that** the refrigerant is CO_2 (carbon dioxide).

7. A vapour-compression-cycle device comprising a compressing device that comprises means for varying the revolutions of the drive shaft of said compressing device, a condenser, at least one cooler, a variable flow control means; and means for controlling said variable control means; and an evaporator, connected in series and forming an integral closed circuit applying a refrigerant operating under trans-critical conditions, **characterized in that** the regulation means for the vapour-compression-cycle device comprises means for regulating the revolutions of the drive shaft of said compressing device for regulating the capacity; and means for controlling said variable control means as a function of the temperature of the refrigerant on the high pressure side with respect to optimizing the COP of the system.

8. A device according to claim 7, **characterized in that** it comprises means for measuring the temperature of the refrigerant on the high-pressure side with respect to optimizing the COP of the system as a function of the temperature or pressure.

9. A device according to claim 7 or 8, **characterized in that** it comprises a control means for regulating the variable control means adapted to optimize the COP as a function of temperature of the refrigerant on the high pressure side for optimizing the COP.

10. Device according to anyone of the claims 7 to 9 **characterized in that** the vapour-compression-cycle device comprises two or more compression stages.

11. Device according to claim 10, **characterized in that** the regulation means for the variable flow control means at decreasing temperature of the refrigerant is arranged to regulate said variable flow control means until a temperature/pressure inferior to the critical temperature is reached, whereupon the opening for refrigerant of the variable flow control means is fully open, and the regulation means at increasing

temperatures starts to regulate said variable flow control means at a temperature higher than the critical temperature.

12. A device according to any one of the claims 10 to 11, **characterized in that** it comprises a first cooler that employs air as coolant arranged in the integral closed circuit after a first compressing stage and before a following compression stage, said first cooler being arranged with respect to the airflow after a second cooler employing air as coolant for cooling/condensing the refrigerant after a second stage compression.

13. A device according to any one of the claims 7 –12, **characterized in that** a cooler that employs water as coolant is arranged in the integral closed circuit after a compressing device and before the air cooled cooler.

14. A device according to one of the claims 7 – 13, **characterized in that** the receiver, in the integral closed circuit is arranged after the variable flow control means but before the economizer.

15. A device according to one of the claims 7 -14, **characterized in that** the operating refrigerant is CO_2 (carbon dioxide).

16. A device according to claim 7-15, **characterized in that** the vapour-compression-cycle device comprises the following devices in the integral circuit. A first compressing device (A), a first cooler (C) employing air as coolant, a second compressing device (B), followed by a counter flow heat exchanger (D) employing water as coolant, and hereafter a second heat exchanger (E) employing air as coolant, after which a variable flow control means (F) is arranged before a receiver (G). In the integral closed circuit after the receiver (G), an economizer (H) is arranged before an expansion vessel (J). Said economizer (H) comprising an inlet connected to the integral closed circuit after the economizer (H)and before the expansion vessel (J), and an outlet connected the integral circuit after the first cooler (C) and before the second compressing device (B). The expansion vessel (J) is connected to the inlet of a

evaporator (K). The outlet of the evaporator (K) is connected to the inlet of the first compressing device (A).

17. A refrigerated container comprising a vapour-compression-cycle device according to claim 7 –16, **characterized in that** the vapour-compression-cycle device comprises coupling coupling to a ship's water cooling system.

18. A refrigerated container comprising a vapour-compression-cycle device **characterized in that** said vapour-compression-cycle device comprises means for coupling to a ship's electrical power system; and that the compression device is driven by an electrical motor means coupled to said electrical power system.

19. Method according to any one of the preceding claims, **characterized in that** it is employed for refrigeration of the interior of a refrigerated container.

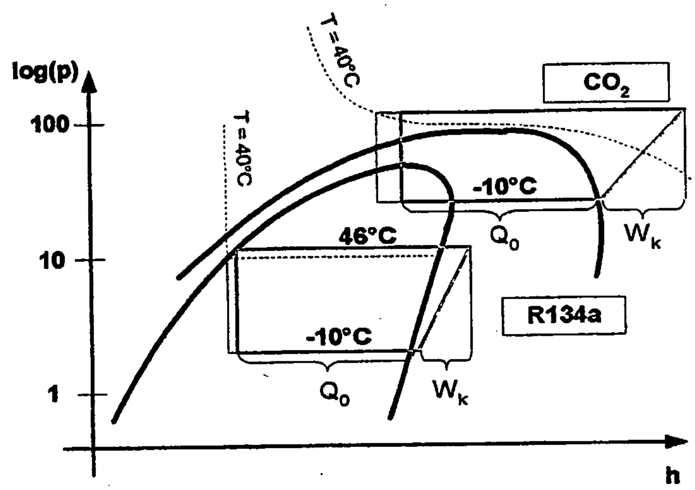

log(p)
100
10
1
h
T=40°C
T=40°C
CO_2
-10°C
Q_0
W_k
46°C
R134a
-10°C
Q_0
W_k

Fig. 1

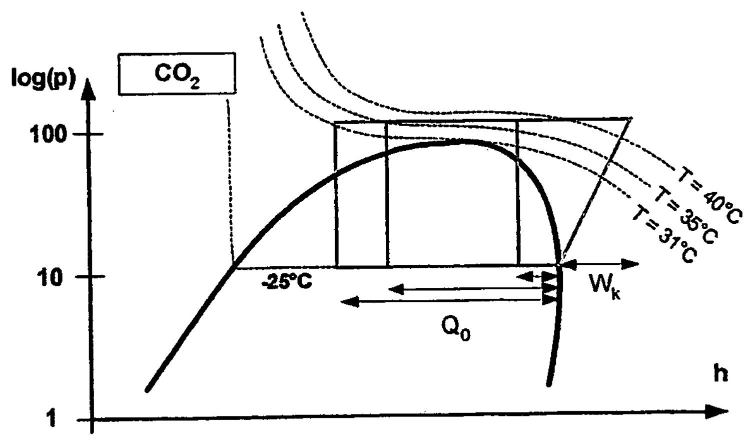
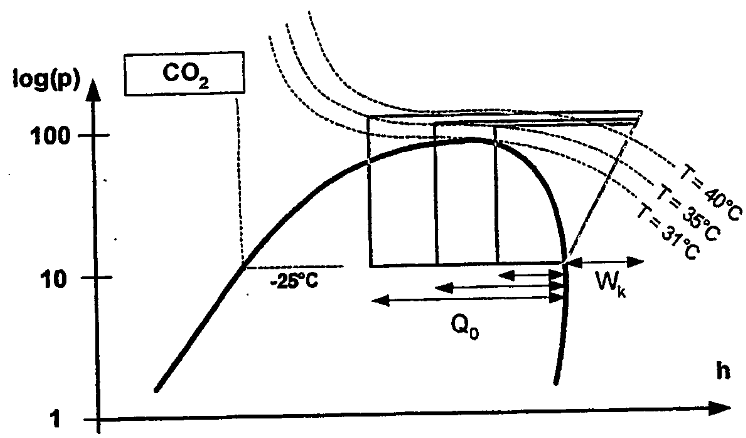
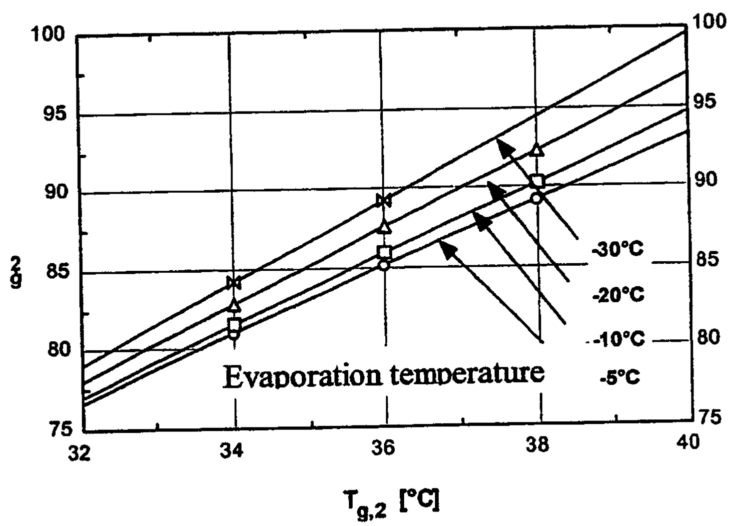

log(p)
CO_2
100
10
1
h
-25°C
Q_0
W_k
T=40°C
T=35°C
T=31°C

Fig. 2



log(p)
CO2
100
10
1
-25°C
T = 40°C
T = 35°C
T = 31°C
Wk
Q0
h

Fig. 3



100
95
90
85
80
75
32
34
36
38
40
-30°C
-20°C
-10°C
-5°C
Evaporation temperature
Tg,2 [°C]

Fig. 4

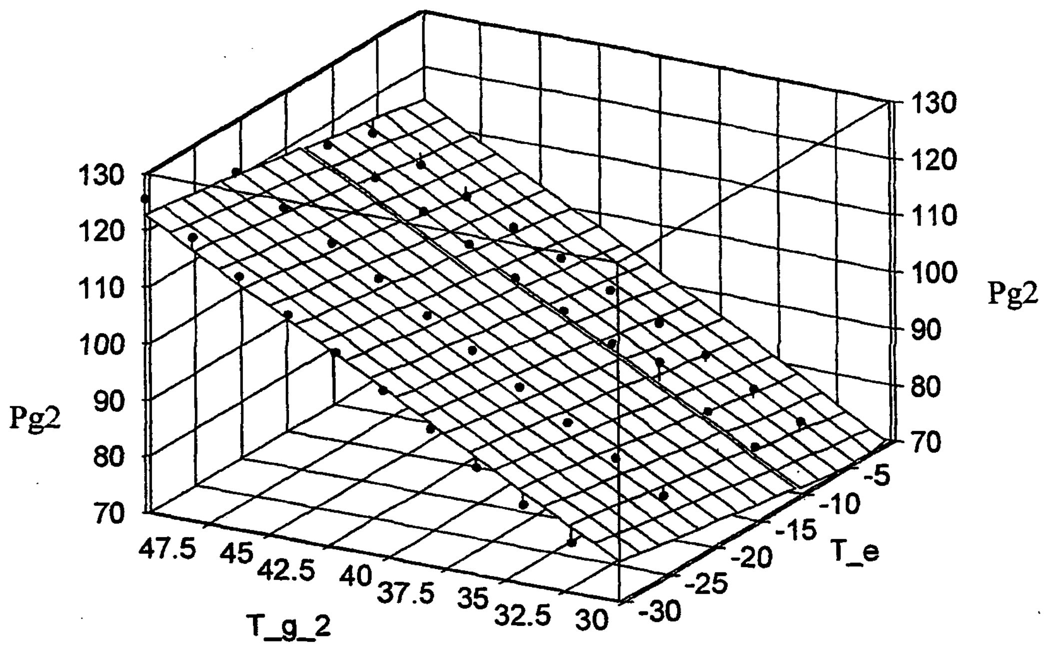

130
120
110
100
90
80
70
Pg2
47.5 45 42.5 40 37.5 35 32.5 30
T_g_2
-5 -10 -15 -20 -25 -30
T_e
130
120
110
100
90
80
70
Pg2

Fig. 5

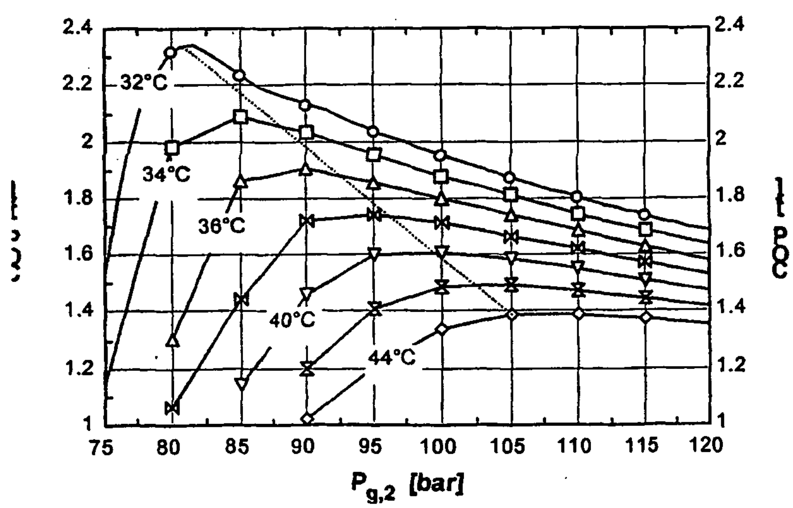

2.4
2.2
2
1.8
1.6
1.4
1.2
1
32°C
34°C
36°C
40°C
44°C
75 80 85 90 95 100 105 110 115 120
$P_{g,2}$ [bar]

Fig. 6

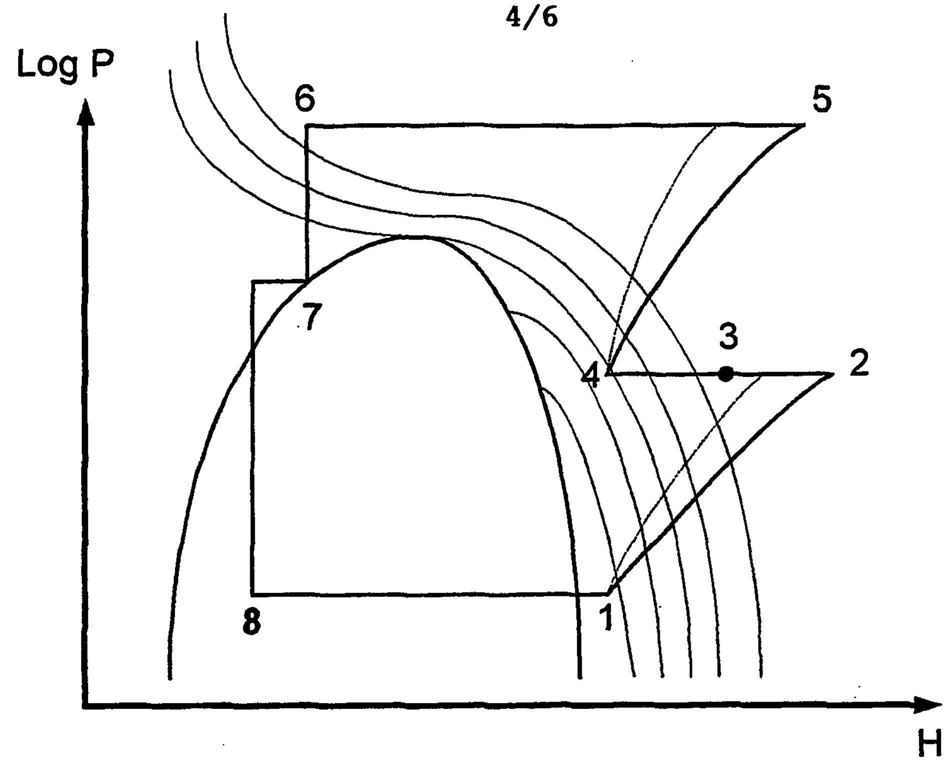


Fig. 7

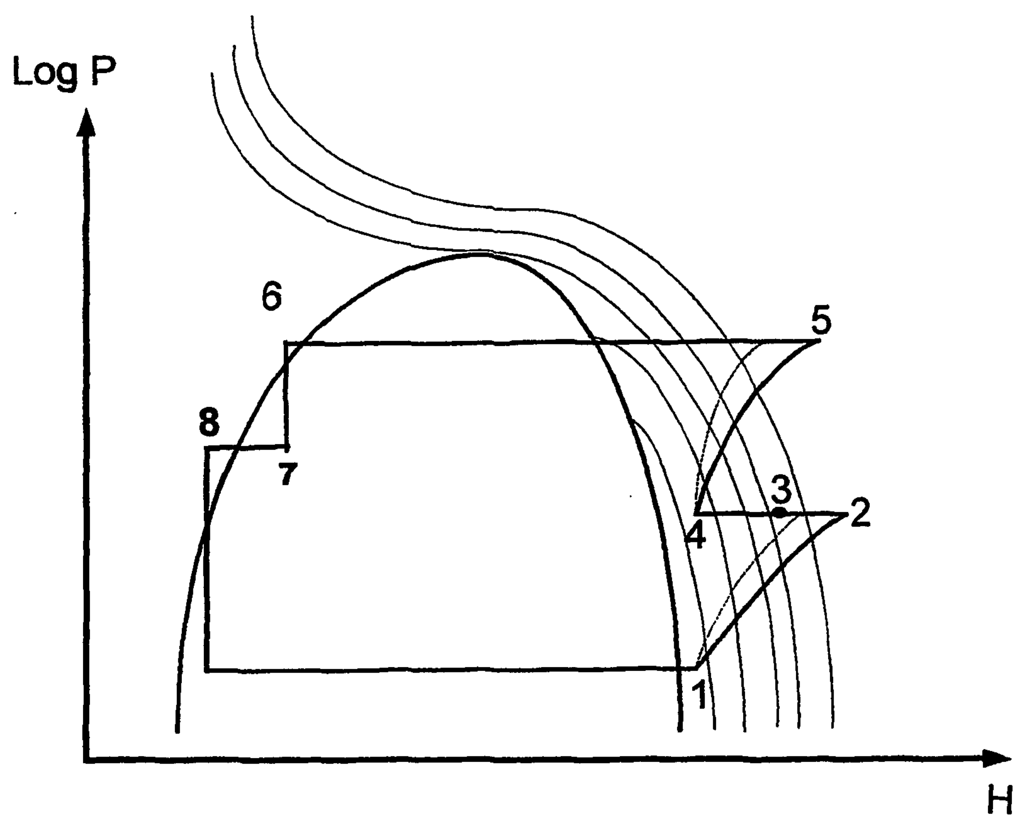


Fig. 8

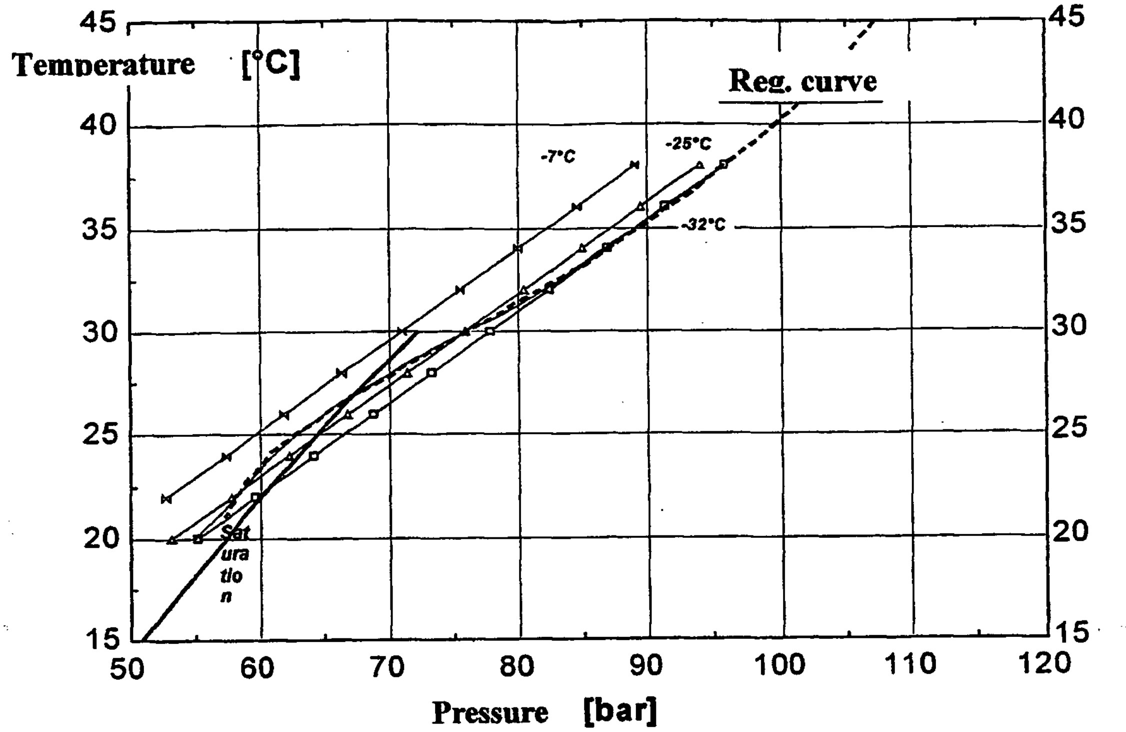

Temperature [°C]
Reg. curve
-7°C
-25°C
-32°C
Sat ura tio n
Pressure [bar]

Fig. 9

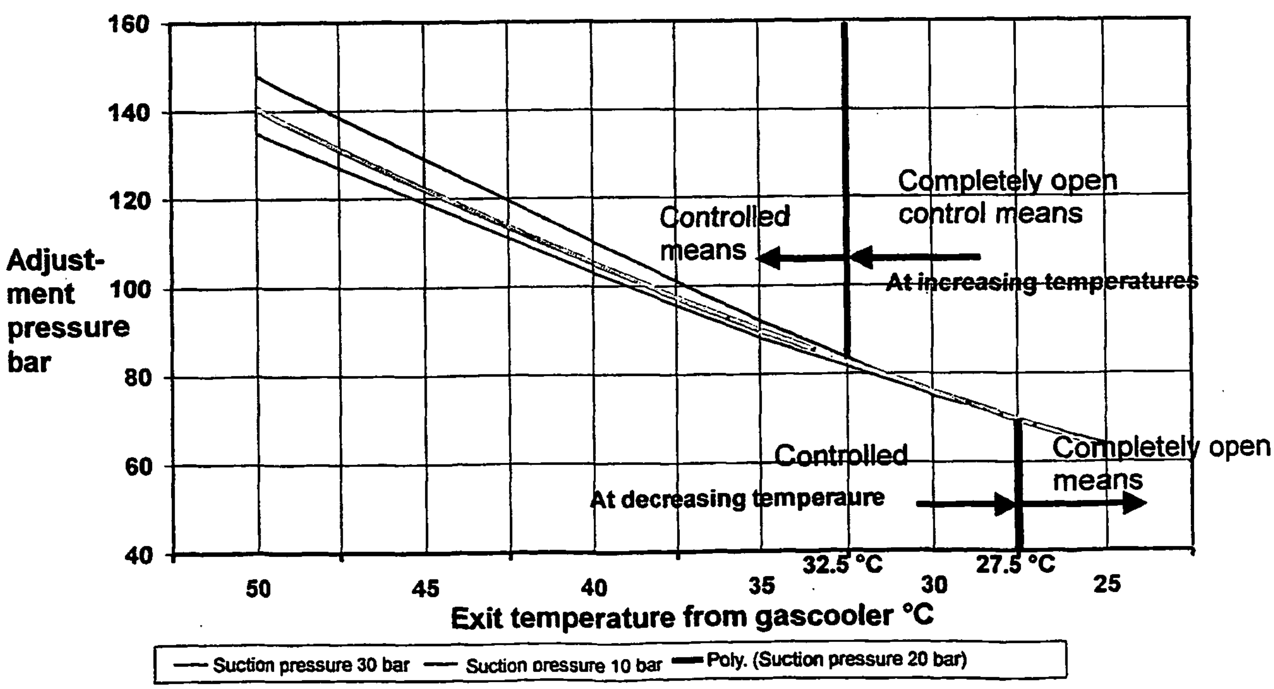

Adjust-ment pressure bar
Completely open control means
Controlled means
At increasing temperatures
Controlled
Completely open means
At decreasing temperaure
32.5 °C
27.5 °C
Exit temperature from gascooler °C
Suction pressure 30 bar
Suction pressure 10 bar
Poly. (Suction pressure 20 bar)

Fig. 10

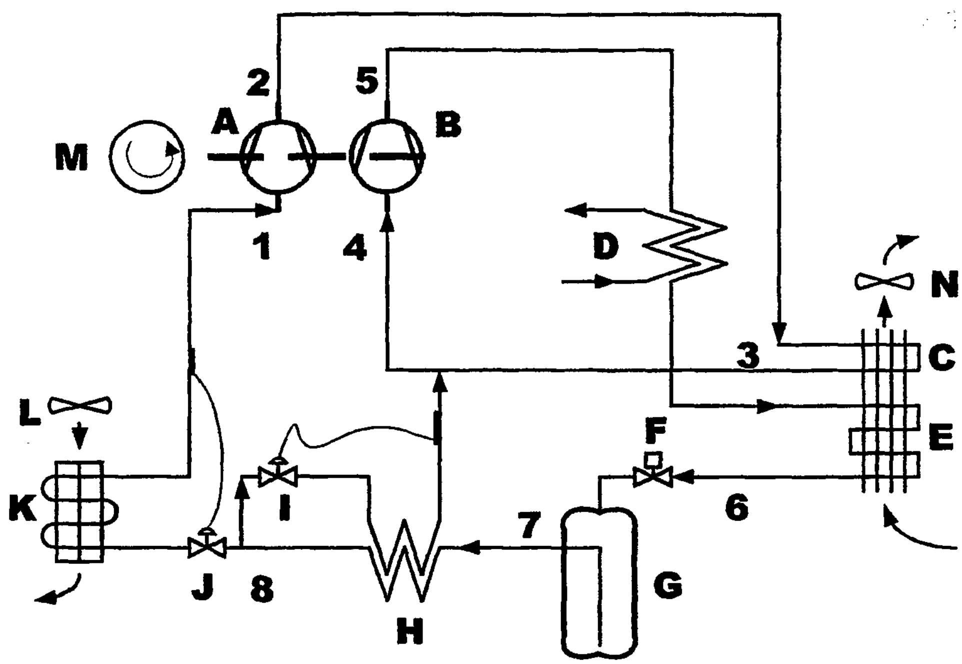


Fig. 11

INTERNATIONAL SEARCH REPORT

International Application No
PCT/DK 02/00570

A. CLASSIFICATION OF SUBJECT MATTER
IPC 7 F25B9/00 F25B49/02

According to International Patent Classification (IPC) or to both national classification and IPC

B. FIELDS SEARCHED

Minimum documentation searched (classification system followed by classification symbols)
IPC 7 F25B

Documentation searched other than minimum documentation to the extent that such documents are included in the fields searched

Electronic data base consulted during the international search (name of data base and, where practical, search terms used)

EPO-Internal

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category °	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
X,P	EP 1 202 004 A (CALSONIC KANSEI CORP) 2 May 2002 (2002-05-02) column 6, line 13-28 column 7, line 39 -column 8, line 20 ---	1,2,7-9, 15
X	US 5 685 160 A (ABERSFELDER GUENTER ET AL) 11 November 1997 (1997-11-11) claims 1,6,15,20; figures 1-7 ---	1,2,7-9, 15
A	US 6 250 099 B1 (KANAI HIROSHI ET AL) 26 June 2001 (2001-06-26) column 1, line 65 -column 3, line 36 ---	1-19
A	EP 1 046 869 A (SANDEN CORP) 25 October 2000 (2000-10-25) page 4, line 14 -page 6, line 51 ---	1-19
	-/--	

[X] Further documents are listed in the continuation of box C.

[X] Patent family members are listed in annex.

° Special categories of cited documents :

"A" document defining the general state of the art which is not considered to be of particular relevance

"E" earlier document but published on or after the international filing date

"L" document which may throw doubts on priority claim(s) or which is cited to establish the publication date of another citation or other special reason (as specified)

"O" document referring to an oral disclosure, use, exhibition or other means

"P" document published prior to the international filing date but later than the priority date claimed

"T" later document published after the international filing date or priority date and not in conflict with the application but cited to understand the principle or theory underlying the invention

"X" document of particular relevance; the claimed invention cannot be considered novel or cannot be considered to involve an inventive step when the document is taken alone

"Y" document of particular relevance; the claimed invention cannot be considered to involve an inventive step when the document is combined with one or more other such documents, such combination being obvious to a person skilled in the art.

"&" document member of the same patent family

Date of the actual completion of the international search	Date of mailing of the international search report
27 November 2002	16. 12. 2002

Name and mailing address of the ISA	Authorized officer
European Patent Office, P.B. 5818 Patentlaan 2 NL - 2280 HV Rijswijk Tel. (+31-70) 340-2040, Tx. 31 651 epo nl, Fax: (+31-70) 340-3016	INGER LÖFVING/JA A

International Application No
PCT/DK 02/00570

C.(Continuation) DOCUMENTS CONSIDERED TO BE RELEVANT

Category °	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
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A	EP 1 059 495 A (MITSUBISHI HEAVY IND LTD) 13 December 2000 (2000-12-13) fig. 1 and adherent text ---	1-19
A	EP 0 786 632 A (DENSO CORP ;NIPPON SOKEN (JP)) 30 July 1997 (1997-07-30) fig. 18 and adherent text -----	1-19

INTERNATIONAL SEARCH REPORT

Information on patent family members

International Application No
PCT/DK 02/00570

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		JP 8110104 A	30-04-1996
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		EP 0952412 A2	27-10-1999
		US 6105380 A	22-08-2000
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		CN 1278052 A	27-12-2000
		EP 1059495 A2	13-12-2000
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		EP 0786632 A2	30-07-1997
		US 5890370 A	06-04-1999